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PULSATION, VIBRATION, AND NOISE ISSUES WITH WET AND DRY SCREW COMPRESSORS

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ABSTRACT

Although wet (oil injected) and dry (oil free) screw compressors are widely used in many applications, limited information is available regarding the pulsation, vibration, and noise problems associated with these types of compressors. This tutorial discusses such problems and includes case histories where field testing was performed, and provides design recommendations.

Both wet and dry screw compressors normally generate pulsation at the pocket passing frequency (PPF) and its multiples. The pulsation amplitudes are affected by many variables such as mole weight of the gas, operating pressures, speed, screw profile, and shape of the discharge port. The pulsation amplitudes are further amplified by acoustical natural frequencies of the compressor/silencer/piping system.

Dry screw compressors are generally supplied with suction and discharge silencers which are designed to attenuate the pulsation generated by the compressors. Most of the silencers are reactive type (Helmholtz filters), absorptive type, or a combination of these two. Although the silencers are designed to attenuate pulsation at certain frequencies, they can also amplify pulsation when the excitation frequencies are coincident with the acoustical natural frequencies of the silencer itself. Pulsation in the silencers can also increase the vibration levels of the compressor rotors and can cause electrical problems to be fed into the local bus.

Therefore, the silencers should be carefully designed to attenuate the pulsation levels over a wide frequency band.

Wet screw compressors typically do not have silencers, since oil separators are used. The oil separators are primarily designed to remove the oil from the gas, but the separators can also attenuate and/or amplify the pulsation generated by the compressor. The shell wall mechanical natural frequencies can also be excited by pulsation at multiples of the PPF, which can result in excessive noise levels and fatigue failures of the shell wall, attached small bore piping, instrumentation, and the oil injection lines.

INTRODUCTION

Screw compressors combine the advantages of positive displacement as for reciprocating equipment, and can provide a wide range of operating conditions. During the past 50 years and especially during the last 30 years, screw compressors have become very popular. When screw compressors were initially being developed, they were generally used to compress air. Today, screw compressors are used to compress a wide range of gases in many different industries including chemical and petrochemical, food processing, pulp and paper, power generation, natural gas, and government/military [11].

Briley [6] provides a good summary of the evolution of screw compressor technology. His paper also discusses the operation and packaging of screw compressors.

In 1975, screw compressors gained more acceptance after the publication of "API Standard 619 (1st Edition) – Rotary Type Positive Displacement Compressors for General Refinery Services". As stated in the API Standard – "*Screw compressors find use in many chemical services. Most significantly, they are employed where no other compressor will work or is economic. They tolerate more abuse than any other comparable unit. On low molecular weight, they enjoy the same benefit as reciprocating compressors since the significance of low density is not detrimental as in the case with centrifugal compressors.*" [1, 16]

During the 1980's, wet screw compressors started to become popular in process gas applications, especially with light gases, such as hydrogen and helium. Some of the other major uses are in refrigeration and in vapor recovery services [17]. In 1984, wet screw compressors were included in the 4th Edition of API Standard 619.

Between 1976 and 2001, several thousand patents were issued for screw compressors. Many of the patents were for various rotor profiles, such as the 3/5, 4/5, 5/6, and 5/7 designs, Figure 1. Although the new designs are in use, the 4/6 design (Figure 1) with 4 lobes on the main (male) rotor and 6 lobes on the gate (female) rotor is by far the most common profile [22].

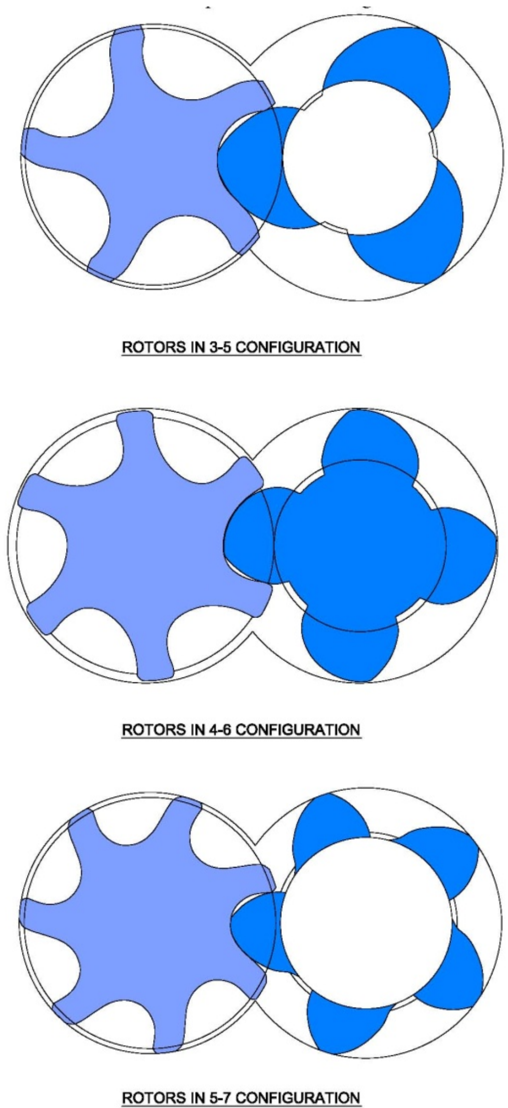


Figure 1: Typical Screw Compressor Rotor Designs

FUNDAMENTALS OF OPERATION

The operation of screw compressors is discussed in many different articles, brochures, text books, and previous Turbomachinery Symposia [13, 16, 17, 22]. (The meshing of the two rotors and the operation of wet screw compressor can be easily visualized in a video developed by Ariel Corporation [20].)

A screw compressor is a positive displacement machine that uses a pair of intermeshing rotors instead of a piston to produce compression. The rotors comprise helical lobes affixed to a shaft. One rotor is called the male rotor and will typically have four lobes. The other rotor is the female rotor and has valleys (flutes) machined into it that match the curvature of the male lobes. Typically the female rotor will have six valleys. For one revolution of the male rotor, the female rotor will only turn through 240 degrees. Since the male rotor is the input shaft, it is said that there are four compression cycles per rotation.

As shown in the Ariel video, the compression process is analogous to a reciprocating compressor. Therefore, it is helpful to refer to the equivalent reciprocating process to visualize how the compression progresses in a screw compressor. Gas is compressed by pure rotary motion of the two intermeshing helical rotors. Gas travels around the outside of the rotors, starting at the top and traveling to the bottom while it is transferred axially from the suction end to the discharge end of the rotor area [18].

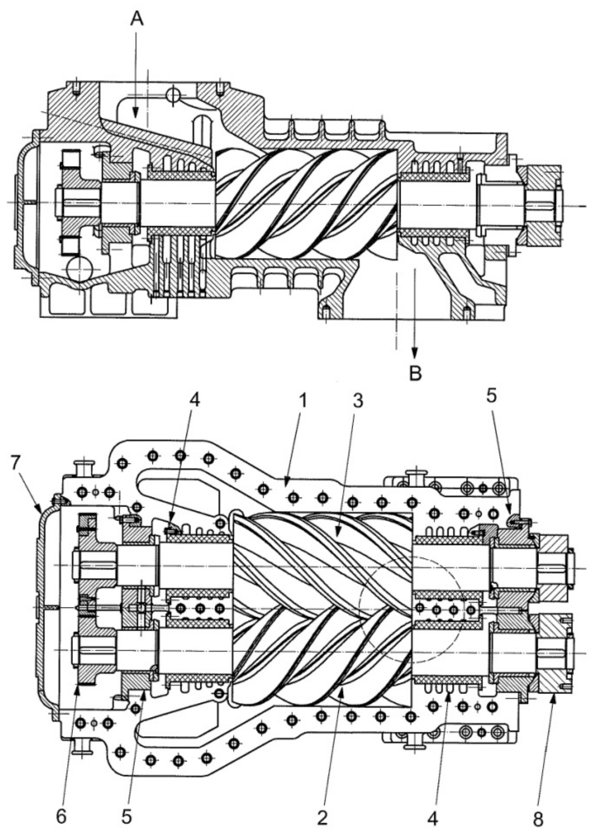
Screw compressors do not have suction or discharge valves like reciprocating compressors; therefore, there is a common misconception that the gas is continuously "extruded" from the compressor, like a sausage grinder, without any significant pulsations. Many sales brochures and instruction manuals suggest that screw compressors do not generate any pulsations and, if they do, the pulsation levels are expected to be very low. Statements that screw compressors do not produce pulsation are in stark contrast to the technical literature. For example, API Standard 619 5th Edition [5] states that "*In screw compressor systems, the flow is not steady, but moves through the piping in a series of flow pulses that are superimposed upon the steady (average) flow*". As discussed later, the pulsation levels are minimized when the compressors are operating near the design conditions, but can be significantly increased at off-design conditions.

TYPES OF SCREW COMPRESSORS

There are basically two types of screw compressors – oil-free machines and oil-injected machines. Although both types use helical rotors, there are several significant differences between the two designs.

Oil-Free Compressors

These compressors are commonly called "Dry Screw Compressors". The male rotor drives the female through synchronizing gears (timing gears) attached to the ends of the rotors. The timing gears prevent contact between the male and female rotors, Figure 2 [5].



Key

A	inlet		
B	outlet		
1	casing	5	radial/thrust bearing
2	male rotor	6	timing gear
3	female rotor	7	end cover
4	shaft seal	8	drive shaft

Figure 2: Dry Screw Compressor (API 619-5th Edition)

The following is a list of major characteristics of dry screw compressors.

1. The process is free of oil; therefore, any gas can be handled.

2. Tight clearances are required between the rotors and between the rotors and the case to reduce leakage and gas blow-by.
3. Seals are required on both shafts.
4. The discharge connections are located vertically on the casing and may be set in an “up-discharge” or “down-discharge” position, Figures 3 and 4. The up-discharge design is commonly used; however, this design can allow broken parts from the silencer to fall back into the compressor when the compressor is not running. Significant damage to the compressor rotors can occur if the broken parts lodge between the two rotors (API 619 1st Edition section 3.5.7.1-d [1] also had a similar statement regarding the up-discharge design – “*Note: This arrangement may result in machine damage in the event of failure of silencer internals.*” This comment was removed from the 2nd Edition and later Editions).

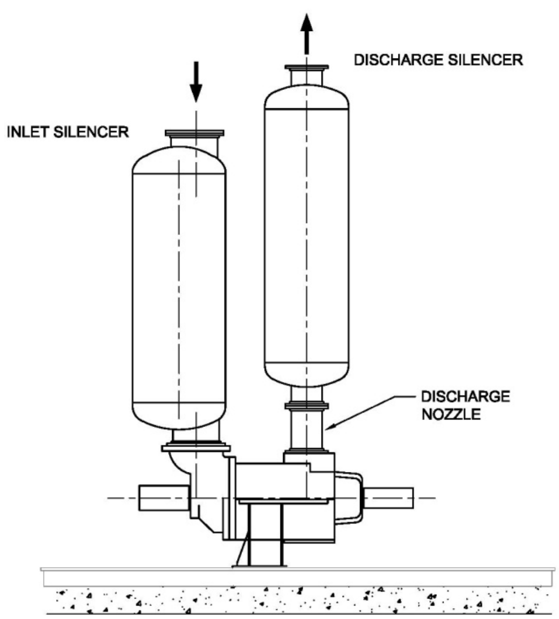


Figure 3: Top (Up) Discharge Design

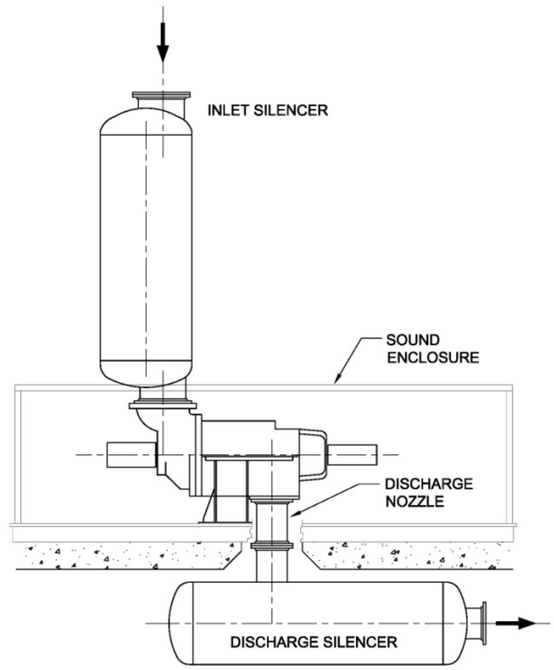


Figure 4: Bottom (Down) Discharge Design

Oil-Injected Compressors

These compressors have several names including “Wet”, “Flooded”, and “Oil Lubricated”. The male rotor drives the female rotor through direct contact – there are no timing gears, Figure 5 [5].

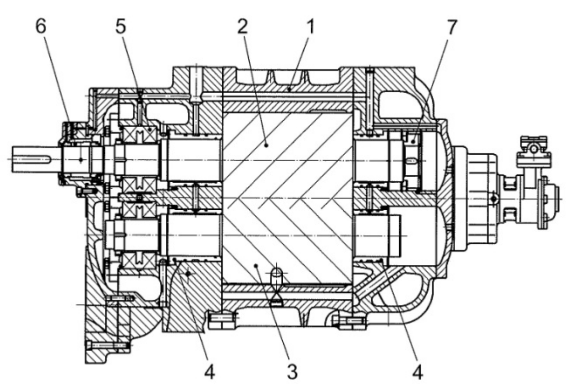
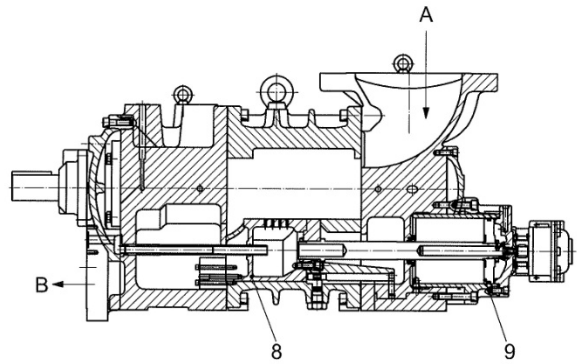


Figure 5a: Wet Screw Compressor (API) 619-5th Edition)



Key

A inlet
B outlet

1 casing
2 male rotor
3 female rotor
4 radial bearing
5 thrust bearing
6 shaft seal
7 hydraulic thrust compensating piston
8 capacity-control slide valve
9 double-acting hydraulic piston

Figure 5b: Wet Screw Compressor (API 619-5th Edition)

The following is a list of major characteristics of wet screw compressors.

1. Oil is injected for lubricating, cooling, and sealing.
2. The clearances between the rotors and between the rotors and case are greater because the oil forms a seal.
3. The running speeds are generally lower because the compressors are more efficient due to less blow-by.
4. The pressure ratios and discharge pressures are significantly higher compared to the dry screw compressors.
5. Power consumption can be greater than dry compressors because of the relatively large amount of oil moving through the system. For example, field data measured by the author on one compressor indicated that the oil flow increased the power by approximately 200 kW out of a total power of 1400 kW.

CAUSES OF PULSATION GENERATION

Both wet and dry screw compressors generate pulsation at multiples of the pocket-passing frequency (PPF), which is defined as the number of lobes on the male rotor multiplied by the compressor running speed in Hz. The maximum generated pulsation levels normally occur at 1x PPF and are generally reduced at the higher harmonics; however, in many cases, higher amplitude pulsation can occur at harmonics of the PPF. It is not uncommon to measure significant pulsation up to the 10^{th} harmonic of the PPF.

Depending upon size and application, running speeds of dry screw compressors can range from approximately 1500 RPM up to 25,000 RPM [22]. Therefore, the pulsations generated by the compressors can occur up to frequencies of several thousand Hz. For example, a compressor operating at 6000 RPM with 4 lobes on the male rotor could generate pulsation from approximately 400 Hz to 4000 Hz (1x PPF to 10x PPF). A similar high-speed compressor operating at 25,000 RPM could generate pulsation from approximately 1,666 Hz to 16,666 Hz.

Wet screw compressors typically operate at slower speeds compared to the dry screw compressors and are often directly coupled to electric motors operating at 1800 RPM (4-pole motor) or 3600 RPM (2-pole motor). For a unit operating at 3600 RPM with 4 lobes on the male rotor, the pulsation frequencies would range from 240 Hz to 2400 Hz (1x PPF to 10x PPF).

The pulsations are generally much higher on the discharge side compared to the suction side. This explains why most noise and vibration problems occur on the discharge sides of screw compressors [14].

The pulsation amplitudes generated by screw compressors are affected by many variables, such as those discussed below.

Shape of Discharge Port

Several authors have explored the influence of various screw compressor design and operating parameters on gas pulsations in the suction and discharge chambers. Mujic and Lovelady [14, 13] reported that changing the shape of a screw compressor discharge port reduces the pulsation levels in the discharge chamber leading to reduced noise levels. Mujic presented an example where it was possible to reduce the noise levels generated by screw compressors by up to 5 dB by reducing the size of the port area. Unfortunately, this reduction was also accompanied by a drop in compressor performance by up to 2% because the modified port required more power input to the compressor for the same working conditions than the original port.

Screw Profile

Although most compressors are manufactured with 4 lobes on the male rotor and 6 lobes on the female rotor, other configurations like the 5/6 and 5/7 and recently the 4/5 and 3/5 are becoming increasingly popular [21]. These other profiles alter the PPF due to the change in the number of lobes on the male rotors. In addition to changing the PPF, the other profiles also affect the delivery rate, efficiency, mechanical strength, manufacturing methods, and cost [22].

Internal Clearances

The internal clearances between the rotors, and the clearances between the rotors and the case, have a significant effect on the pulsation levels generated by the compressor. In many cases, the pulsation levels can be significantly increased when these clearances are tight. This explains why the pulsation, vibration, and noise levels are oftentimes increased after the compressor has been overhauled to restore the clearances to the original design values. These effects are also discussed in Case History No. 3.

Internal Volume Ratio

In a reciprocating compressor, the discharge valves open when the pressure in the cylinder exceeds the pressure in the discharge manifold. Because a screw compressor does not have valves, the angular positions of the rotors determine the locations of the discharge ports which determine the maximum discharge pressure that will be achieved in the screw threads before the compressed gas is pushed into the discharge piping [18, 22].

The built-in or internal volume ratio is the fundamental design characteristic of all screw compressors and is determined by the casing geometry. The compressor is a volume reduction device. The comparison of the volume of trapped gas at suction (V_s), to the volume of trapped gas remaining in the compression chamber when it opens to discharge (V_d) defines the internal volume reduction ratio of the compressor.

The volume index or "V_i" determines the internal pressure ratio (P_i) of the compressor and the relationship between them can be approximated as follows [18].

$$V_i = V_s / V_d \qquad (1)$$

where,

V_i = internal volume ratio

V_s = internal suction volume, acf

V_d = internal discharge volume, acf

The internal pressure ratio is also defined as the ratio of the discharge and suction absolute pressures.

$$P_i = P_d / P_s \qquad (2)$$

where,

P_i = internal pressure ratio

P_d = internal discharge pressure, psia

P_s = internal suction pressure, psia

The relationship between the built-in pressure ratio (compression ratio) and the internal volume ratio is as follows:

$$P_i = V_i^k \ \text{ or } \ V_i = P_i^{1/k} \qquad (3)$$

where,

P_i = internal pressure ratio

k = specific heat ratio of the gas being compressed, typically 1.26 – 1.3

k = c_p/c_v for ideal gases

Only the suction pressure and the internal volume ratio determine the internal pressure level in the trapped pocket before opening to the discharge port. However, the system determines the discharge pressure in the discharge piping.

The discharge pulsation levels downstream of the compressor are directly controlled by the difference between the compressor internal pressure and the pressure in the discharge piping. The discharge pulsation levels are minimized when the internal pressure is approximately equal to the discharge line pressure.

Well Suited Built-In Volume Ratio

An example of a pressure-volume diagram for a screw compressor with a well suited built-in volume ratio is shown below in Figure 6 [18]. P_{dl} refers to the discharge line pressure.

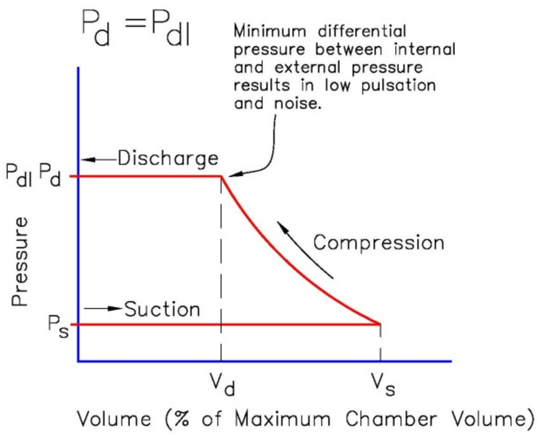


Figure 6: Pressure-Volume Diagram V_i Matches Operating Conditions

Over-Compression

If the internal volume ratio of the compressor is too high for a given set of operating conditions, the pressure of the internal gas will be raised above the discharge pressure in the piping. This is called over-compression and is illustrated in the pressure-volume curve shown in Figure 7 [18].

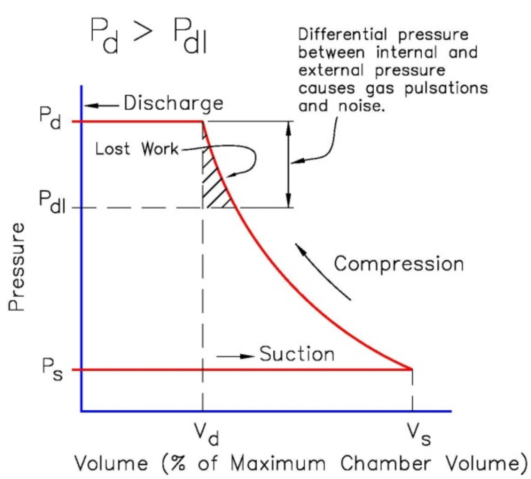


Figure 7: Pressure-Volume Diagram Over-Compression, V_i Too High

In this case, the higher pressure gas in the screw thread expands out of the compressor into the discharge resulting in significant flow modulation (pulsation) in the discharge line.

Under-Compression

When the compressor volume ratio is too low for the system operating pressures, under-compression occurs as shown in the sketch in Figure 8 [18].

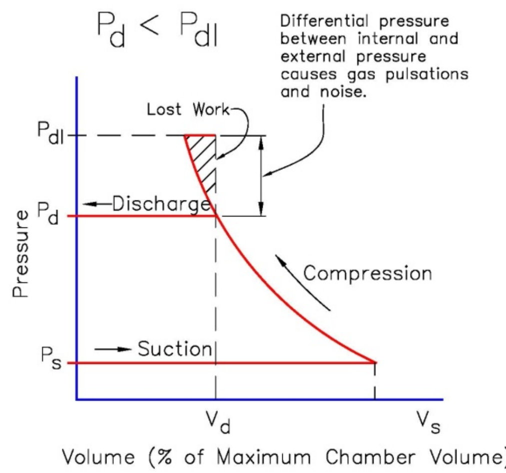


Figure 8: Pressure-Volume Diagram Under-Compression, V_i Too Low

In this case, the discharge port opening occurs before the internal pressure in the compressor trapped pocket has reached the system discharge pressure level. The higher pressure gas outside the compressor flows back into the lower pressure pocket, creating flow modulation (pulsation) in the discharge line.

In both cases (over-compression and under-compression), the compressor will still function, and the same volume of gas will be moved, but more power will be required than if the discharge ports are correctly located to match the compressor volume ratio to what the system needs. Perhaps more importantly, the discharge pulsation levels will be significantly increased.

Typical Internal Volume Ratios

A low V_i compressor corresponds to a low compression ratio machine. Similarly, high V_i compressors are used on higher compression ratio systems.

Screw compressor manufacturers usually offer different volume ratio machines. The most common V_i range is 2.2 to 5.0. There are some machines offered outside this range for special applications, but this is the most common. The following table provides a comparison of compression ratios and their corresponding ideal volume ratios based on the above formulas (calculated with "k" value = 1.3) [7].

Compression Ratios and Corresponding Volume Ratios

Compression Ratio	Ideal V_i Ratio
2.0	1.7
2.5	2.0
3.0	2.3
3.5	2.6
4.0	2.9
4.5	3.2
5.0	3.4
5.5	3.7
6.0	4.0
6.5	4.2
7.0	4.5
7.5	4.7
8.1	5.0
8.5	5.2
9.0	5.4
9.5	5.7
10.0	5.9

As discussed above, when selecting compressors, it is very important that the compressor volume ratio match the system operating conditions. For example, if a high V_i compressor is operated on a low compression ratio application, over-pressure will occur. Similarly, if a low V_i compressor is operated in a high ratio application, under-pressure will occur.

When designing a system with a fixed volume compressor, the V_i should be carefully selected. Consideration should be given to the system operation during the entire life cycle, taking into consideration the expected range of operating conditions and system pressures [8]. Due to the limited number of available compressor designs, user companies often purchase compressors that have incorrect volume ratios which often results in systems with high pulsation, vibration, and noise levels.

Variable V_i Compressors

The volume ratio of a dry screw compressor usually cannot be changed without major modifications to the compressor. However, the volume ratio of a wet screw compressor can usually be changed by moving the slide valve/slide stop assembly, or by replacing the slide valve assembly with one which results in a different volume ratio.

The slide valve is normally used to vary the capacity of the compressor by recirculating gas back to the suction side of the compressor. The slide valve can also be used to change the internal ratio to match the operating conditions, such as when the pressures, molecular weights, or temperatures of the gas are changed.

On machines with "Variable V_i", the slide valve assembly is moved along the axis of the rotors to change the volume ratio by making an external adjustment on the compressor. On some machines, the volume ratio can be changed while the compressor is in operation and some machines can vary the volume ratio automatically.

For "Fixed V_i" machines, some manufacturers offer several different slide valves to change the volume ratio for various operating requirements. The replacement of these slide valves usually requires disassembly of the compressor.

The operation of the slide valve with regard to changing the volume ratio is explained in more detail in the Ariel movie [20]. Note that dry gas compressors usually do not have slide valves, especially large high-capacity units, because special lubrication and/or flushing of the slide valve in its guide channel may be required to prevent the slide valve from freezing in one position.

Molecular Weight of the Gas

The acoustical natural frequencies of the compressor/silencer/piping system are controlled by the speed of sound of the gas which is a direct function of the molecular weight of the gas. Many times, a compressor system will operate with low pulsation, vibration, and noise levels with a particular gas but cannot operate with other gases due to the changes in the acoustical natural frequencies caused by the change in molecular weights. These effects are discussed in Case Histories No. 2 and No. 3.

Dry screw compressors can experience pulsation, vibration, and noise problems during the initial startups since they are often started with air, nitrogen, and ultimately the actual process gas. The molecular weight of the process gas is often significantly different than the molecular weight of the air and nitrogen.

Many papers and brochures state that due to the positive displacement characteristics, screw compressors are relatively insensitive to changes in the molecular weights of the gas and smoothly handle gases with varying molecular weights, pressures, and temperatures [16]. These statements consider only the compression of the gas, and do not agree with the author's experience obtained on many field tests of problem compressors, in that the pulsation, vibration, and noise levels can change significantly due to relatively minor changes in the molecular weight of the gas.

Operating Pressures

The discharge pulsation levels are generally increased at higher operating pressures because the gas densities are increased. The higher densities will also change the speed of sound of the gas which, as previously stated, will also change the acoustic natural frequencies of the system.

Running Speed

The pulsation levels are also increased at higher running speeds.

PULSATION-INDUCED VIBRATION

Pulsation can interact directly or indirectly to produce high vibration of the compressor rotors, case, silencers, main piping, small-bore piping, and instruments. The high frequency content not only allows for excitation of directly attached piping, but also can produce structure borne energy that can excite components of unrelated but nearby systems.

Additional amplification can occur when components are mechanically resonant with the pulsation frequencies. When the pulsation mode shapes are similar to the structural mode shapes, efficient coupling mechanisms exist that will cause excessive vibration and noise. However, even when the mode shapes are dissimilar, pulsation will often still couple efficiently to produce structural vibration and noise.

Rotor Vibration

Pulsation in the compressor discharge port at the PPF can cause the compressor rotor vibration levels to be increased. The rotor vibration levels can be further increased if one of the rotor lateral natural frequencies is near the pulsation frequency. In fact, high rotor vibration levels are often the first indication of excessive pulsation levels in the compressor discharge port. In some cases, the high pulsation levels can cause the rotor vibration levels to be increased above the allowable levels.

Compressor Case Vibration

The pulsations internal to the compressor case can also cause excessive vibration of the compressor case which can result in excessive noise levels. On dry screw compressors, the noise levels can be further increased by vibration of the gearbox housing.

Small Bore Piping Vibration

Pulsation can also result in increased vibration and fatigue failures of small-bore piping and instrumentation attached to the discharge silencer and to the piping downstream of the silencer. The vibration levels can be further increased if the pulsation frequency is coincident with one of the mechanical natural frequencies of the small bore piping or the instrumentation.

Shell Wall Vibration

The pulsation can cause the shell wall vibration levels to be increased on the silencers of dry screw compressors, the oil separators of wet screw compressors, and the piping downstream of the silencers. The shell wall vibration of the vessels and piping can similarly be increased if the pulsation frequency is coincident with a shell wall or lateral (length mode) mechanical natural frequency. Excessive shell vibration can result in fatigue failures of the piping and very high noise levels.

The shell wall thickness of most silencers, separators, and downstream piping are often designed based upon the static pressure and the corrosion requirements. It is recommended that the wall thickness of the vessels and the piping be increased to reduce the shell wall vibration and the resulting noise levels. The initial capital costs for the thicker wall thickness may be higher but it will be worth it in the long term.

In addition, when dealing with corrosive gases, the wall thickness should not be reduced when using stainless steel or other non-carbon-steel material. API Standard 619 Section 6.9.9 states – "*The thickness for non-carbon-steel shell material shall be equal to or greater than the thickness required for carbon-steel, including the carbon-steel corrosion allowance.*" [5], this problem is discussed in Case History No. 3.

PULSATION ATTENUATION

Orifice Plates

Orifice plates are often effective in attenuating pulsations generated by the compressor. The addition of orifice plates is generally considered to be a fairly simple modification and numerous pulsation, vibration, and noise problems have been solved simply by installing orifice plates.

Orifices attenuate the pulsation levels by adding acoustical resistance (damping) to the system. A common misconception is that orifice plates shift acoustical natural frequencies away from excitation frequencies; however, this is not correct because the pressure drop across the orifice is not sufficient to change the acoustical boundary conditions.

Orifice plates are effective in attenuating the pulsation levels at certain acoustical natural frequencies; however, the addition of an orifice plate will not attenuate the pulsations at all of the multiples of the PPF. In order for the orifice plates to have maximum effectiveness in attenuating the pulsation at a particular acoustic natural frequency, the orifices must be installed at a location which corresponds to the point of maximum particle velocity for that particular acoustical natural frequency.

For example, orifice plates are often installed to attenuate the pulsations at the compressor discharge nozzle acoustical natural frequencies. The acoustical natural frequencies of the discharge nozzle can be approximated by assuming that the discharge nozzle is a ¼ wave length resonator (¼ wave stub) with an open end at the discharge silencer and a closed end at the rotor discharge port.

The optimum location for the orifice is at the location of maximum particle velocity which is the open end of the nozzle; however, it is generally difficult to install an orifice at this location without welding the orifice plate into the nozzle, Figure 9.

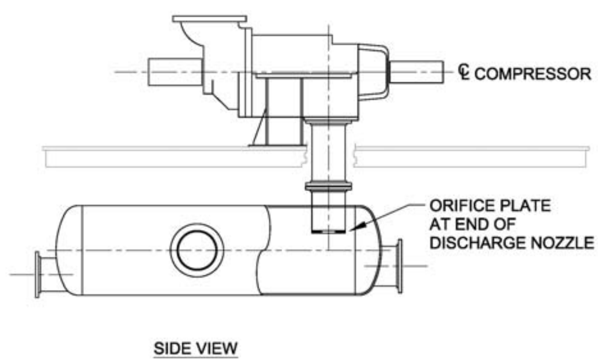


Figure 9: Orifice Plate Installed at the End of the Discharge Nozzle

Therefore, the orifice plate is generally installed in the discharge nozzle flange which will be a few inches from the open end, Figure 10.

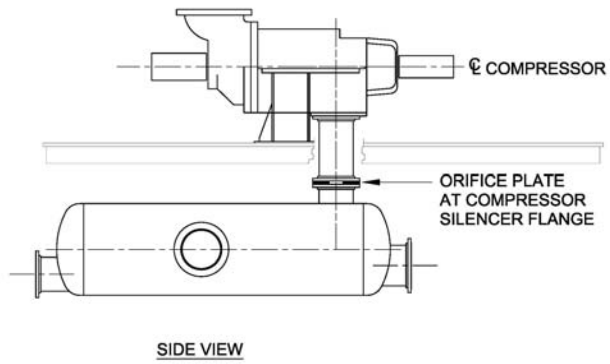


Figure 10: Orifice Plate Installed at the Silencer Flange

Since the compressor discharge flange is not the optimum location for the orifice, additional pressure drop may be required to attenuate the pulsations to the desired level.

For screw compressors, the typical pressure drop across the orifice plate required to sufficiently attenuate the pulsation levels could be as high as 2 – 4 percent of the discharge line pressure. The API Standard 619 1st Edition [1] originally limited the allowable pressure drop through the silencer to 1 percent of the absolute pressure times the specific gravity of the gas, which in many cases results in minimal allowable pressure drop. The allowable pressure drop was later increased in the current API Standard 619 5th Edition [5] which states that "*the pressure drop through the discharge pulsation suppressors/silencers shall not exceed 2.5 percent of the absolute pressure*". The API 619 allowable pressure drop could be exceeded if the orifice plate has a pressure drop of 2 percent by itself. In this case, an exemption from the API 619 specification might be required.

The pulsation levels are normally reduced in a fairly linear manner as the pressure drop across the orifice is increased; however, at some point, the system becomes non-linear and the amount of pressure drop required to further attenuate the pulsation is significantly increased. Therefore, computer simulation or field experimentation with different sized orifices is often required to determine the optimum pressure drop.

The thickness of orifice plates used with reciprocating compressors is typically 3/16 inches to 3/8 inches (0.187 – 0.375 inches). However, the author recommends that the orifice thickness on screw compressors should be increased to approximately 0.5 – 1.0 inches to avoid possible fatigue failures (this problem is discussed in Case History No. 1). For thick orifice plates, the typical orifice pressure drop calculations often over-estimate the amount of pressure drop. Therefore, when computing the pressure drops with thick orifice plates, the author recommends using the equations for computing the pressure drops for chokes. The orifice plates can have a single hole or multiple holes. The pressure drops for a multi-hole orifice can be estimated by considering the total area of the holes.

Orifice plates are effective in attenuating pulsations; however, the increased pressure drop also increases the power costs. One user company was successful in eliminating their screw compressor noise and vibration problems by installing orifice plates with high pressure drops of approximately 10 percent of the discharge pressure; however, the electrical power costs were excessive. The final solution was to replace the original discharge silencers with new silencers which attenuated the pulsations without excessive pressure drop.

Pulsation Dampeners

As discussed above, API Standard 619 recommends that, "*Unless otherwise specified, inlet and exhaust pulsation suppressors/silencers for each casing shall be supplied by the compressor manufacturer. Their primary function shall be to provide the maximum practical reduction of pulsations in the frequency range of audible sound without exceeding the pressure drop limit specified*" [5].

The term "silencer" is actually a misnomer. A silencer should actually be referred to as a pulsation attenuation device or dampener as the purpose of the silencer is to reduce pulsation in the downstream piping. Since significant noise levels can be created due to the vibration of the silencer shell wall, particularly when the pulsation frequencies are coincident with one or more of the shell wall natural frequencies, the silencer can actually act as a noise amplifier.

Dry screw compressors are generally supplied with silencers on both the suction and discharge sides. This practice could be a “carry-over” from the design of reciprocating compressors which always have pulsation dampeners on both the suction and discharge sides. Pulsation dampeners are required on the suction side of reciprocating compressors because the suction pulsation levels are significant and can result in excessive piping vibration levels if untreated. The suction pulsation levels on screw compressors are generally very low. The author's experience is that noise or vibration problems are nearly always associated with the discharge side of dry screw compressors which would suggest that silencers might not be required on the suction side of dry screw compressors.

Three types of silencers are typically employed – absorptive, reactive, and combination (reactive/absorptive). The characteristics of these silencers are discussed below.

Absorptive Type

These silencers are also referred to as dissipative silencers. This design depends on sound absorbing material to dissipate the sound energy. The sound waves pass through the spaces between the tightly packed, small diameter fibers of the absorptive material and the resulting viscous friction dissipates the sound energy as small amounts of heat [15].

The absorptive silencers are similar to “glass-pack” mufflers used on cars. These designs provide good attenuation for high-frequency pulsation but less attenuation at the low and middle frequencies. In cars, the higher frequencies are attenuated leaving the lower frequency “rumbling” type noise which is often favored by “hot rod” car enthusiasts.

The absorptive material (glass wool, steel wool, foam) is contained along the inside edges of the vessel and often in a tube along the center axis, Figure 11.

Figure 11: Absorptive Silencer

The absorptive material is generally restrained with perforated steel sheets, or steel mesh. The restraining materials often experience fatigue failures which release the absorptive material into the downstream piping. The perforated sheets and the mesh can also pass into the downstream piping and into the compressor on up-discharge designs.

Absorptive silencers cannot be used in some processes due to contamination of the absorptive material from liquids entrained in the gas and possible chemical reactions which can form solids, etc.

Reactive Type

These silencers, also referred to as Helmholtz filters, are multi-chamber designs which are effective in reducing low-frequency pulsation; however, these designs are generally not effective in attenuating high-frequency energy at the higher multiples of the PPF. These types of filters are commonly used on reciprocating compressors to attenuate low-frequency pulsation.

Typical pulsation filter designs consist of two or three chambers connected by smaller diameter choke tubes, Figures 12 and 13.

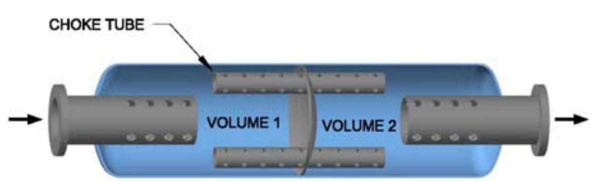


Figure 12: Reactive Silencer - Two Chambers

These reactive filters are referred to as volume-choke-volume filters and are designed to attenuate the fundamental pocket-passing frequency. The combination of these acoustic elements produces a low-pass filter which attenuates pulsation above the cutoff frequency.

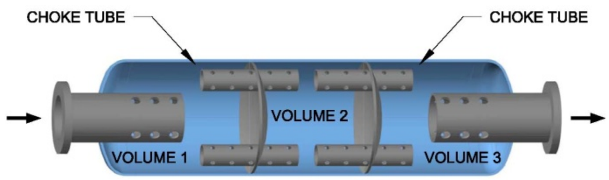


Figure 13: Reactive Silencer - Three Chambers

The cutoff frequency is the resonant frequency of the filter which is also called the Helmholtz frequency. The pulsations will be amplified at the Helmholtz frequency; therefore, the filter is designed with the Helmholtz frequency well below the primary excitation frequency which in this case is the PPF.

Reactive silencers have internal acoustical natural frequencies which can amplify the pulsation generated by the compressor when the excitation frequencies are coincident with these acoustical natural frequencies. The internal acoustical natural frequencies include: the inlet nozzle resonance (1/4 wave length modes), choke tube resonances (1/2 wave length modes), chamber length resonances (1/2 wave length modes), and cross-wall resonances (across the diameters of the chambers). The acoustical natural frequencies are referred to as passbands because the pulsations are not attenuated at these frequencies and can pass into the downstream piping. These acoustical natural frequencies are discussed in more detail in a paper previously presented by the author at the 28th Turbomachinery Symposium [19].

Combination Type (Reactive and Absorptive)

This silencer type is a multi-chamber reactive silencer with absorptive material, such as that shown in Figure 14.

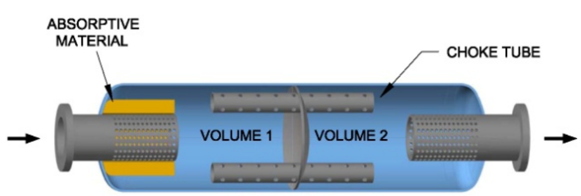


Figure 14: Combination Silencer - Two Chambers with Absorptive Material

There are also many versions of this design and some configurations have additional absorptive material along the sides of the chambers.

With the typical silencer designs, it is unlikely that a silencer can be designed which can attenuate the excitation energy without also amplifying some of the excitation frequencies. In addition, the silencer has to be able to withstand the high-frequency energy without experiencing fatigue failures of the internal parts and the vessel itself.

Therefore, it is recommended that detailed acoustical analysis be performed to verify that the system will perform satisfactorily with acceptable pulsation, vibration, and noise levels. API 619 5th Edition Section 6.9.1 [5] states – "*The requirement for, and the scope of, an analysis of pulsation and noise suppression shall be agreed between the purchaser and the vendor.*" Previous Editions [1, 2, 3, 4] stated that – "When specified, the pulsation suppressor/silencer vendor shall supply detailed drawings to permit an independent study of the acoustical characteristics of the pulsation suppressor/silencers together with the purchaser's piping system".

Oil Separators

Wet screw compressors incorporate a pressurized reservoir and a gas/oil separator. The oil separator is a specialized piece of equipment that often includes proprietary internal design features. It is designed to effectively remove the oil entrained in the process-gas stream prior to final process-gas discharge from the package [5].

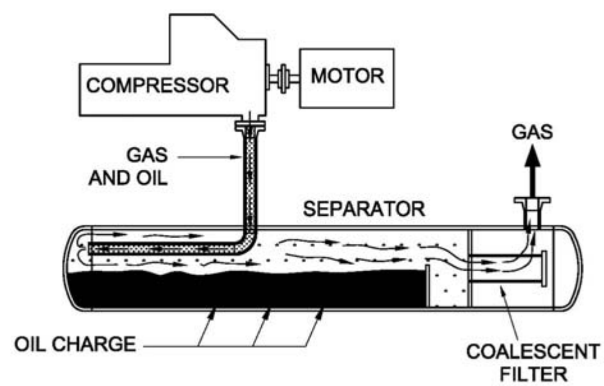


Figure 15: Horizontal Separator

Oil separators can be horizontal or vertical. Horizontal separators are usually installed below the compressor (Figure 15) and vertical separators are installed downstream of the compressor, Figure 16.

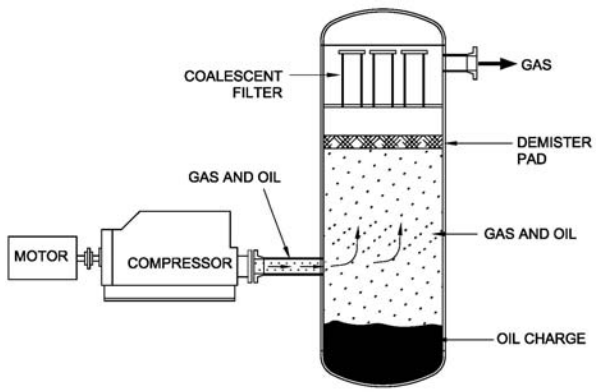


Figure 16: Vertical Separator

The oil separation process is similar with both designs in that the mixture of discharge gas and oil leaving the compressor is directed against one side of the oil separator where it experiences a change in direction and a large reduction in velocity. The larger oil particles are drawn to the oil sump by gravity, with the smallest particles carried into the coalescing filters. Here, these small oil particles impact on the internal fibers in the filters and coalesce into larger oil droplets which can then be collected in the dry end sump and returned back to the compressor [18].

The oil separators are acoustically similar to reactive silencers for dry screw compressors. The configuration with the two chambers connected by coalescing tubes creates a volume-choke-volume Helmholtz filter which attenuates the pulsation levels. The pulsation levels are much higher in the first chamber compared to the second chamber downstream of the coalescing filters. The absorptive fibers in the coalescing filters can be effective in attenuating the high-frequency pulsation. One separator designer reported that the pulsation in the first chamber could be similarly attenuated by installing absorptive material in the separator inlet nozzle (like the combination type silencer used with dry screw compressors).

The separators are also similar to the reactive silencers in that the pulsations generated by the compressor can be amplified by acoustical natural frequencies of the discharge nozzle and cross-wall acoustical natural frequencies of the vessel. The pulsations can also excite the shell wall natural frequencies of the vessel resulting in high vibration and excessive noise levels. In some cases, the high vibration can result in fatigue failures of the vessel and small bore piping. These problems are discussed in Case Histories No. 5 and 6.

NOISE PROBLEMS

Screw compressors are known to generate high level noise and double-hearing protection (ear plugs and ear muffs) is often required for personnel working near the compressors.

API Standard 619 does not provide an allowable sound pressure level. The 5th Edition Section 5.1.19 [5] states – "*Control of the sound pressure level (SPL) of all equipment furnished shall be a joint effort of the purchaser and the vendor having unit responsibility. The equipment furnished by the vendor shall conform to the maximum allowable sound pressure level specified. In order to determine compliance, the vendor shall provide both maximum sound pressure and sound power level data per octave band for the equipment. Note: The sound power level of a source can be treated as a property of that source under a given set of operating conditions. The sound pressure level, however, varies depending on the environment in which the source is located as well as the distance from the source. Vendors routinely take exception to guaranteeing a purchaser's maximum allowable sound pressure level requirement due to the argument that the vendor has no control over the environment in which the equipment will be installed. The vendor has control, however, over the sound power level of the equipment.*"

Oftentimes, purchasers specify an allowable sound pressure level of 85 or 90 dBA measured 3 ft. from the compressor. The author considers that these allowable values are unrealistic and should be increased to 90 – 95 dBA because the background levels in a plant are usually above 85 dBA and the lower levels are very difficult (if not impossible) to achieve. Since it is almost impossible to achieve the purchasers' allowable levels, the noise levels near the screw compressors are often disregarded and signs are posted warning that the noise levels are excessive and double-hearing protection is required.

As stated in API 619 Section 5.1.20 – "*These compressors tend to be very noisy*". In an effort to reduce the noise levels, dry screw compressors are often installed in sound enclosures. The enclosures enclose the compressor and the gearbox; however, the suction and discharge silencers are often outside the enclosure. Although the sound enclosures can reduce the noise levels generated by the vibrations of the compressor and the gearbox, the enclosures are expensive and can cause the temperatures inside the enclosure to be increased resulting in cooling problems in hot climates.

A significant amount of the noise near screw compressors is due to shell wall vibration of the discharge silencers, downstream piping, and oil separators. The shell wall vibration creates sound waves similar to those generated by a large speaker. Therefore, in order to reduce the noise (sound pressure levels), it is important to minimize the shell wall vibration and to install acoustical lagging on the silencers, piping, and oil separators.

DYNAMIC-INSERTION LOSS

API 619 5th Edition [5] states that the silencer manufacturer should provide information on the dynamic-insertion losses for each octave band. The current Edition does not provide a specific value for the insertion loss; however, one of the previous Editions stated that the discharge silencers should have a minimum noise insertion loss of 30 dB and the inlet silencers should have a minimum noise insertion loss of 20 dB.

The following terms – *Transmission Loss* (TL), *Dynamic Insertion Loss* (DIL), and *Noise Reduction* (NR) are sometimes used by silencer manufacturers to refer to the difference between the sound pressure level measured at the inlet of a silencer and at its outlet [10]. Transmission Loss is defined as the ratio of incident power to transmitted power. Dynamic Insertion Loss is defined as the ratio of sound pressure without a silencer installed to the sound pressure with a silencer at a downstream location. Noise Reduction is defined as the ratio of upstream sound pressure to downstream sound pressure. All of these quantities are expressed using the dB scale.

Both TL and NR are measured, in general, with the silencer installed. Calculated or measured DIL and NR values can be changed even with an identical silencer depending on upstream and downstream acoustical boundary conditions, or on measurement locations unless infinite duct (or no reflective boundary condition) is used in both upstream and downstream. TL value can be mildly dependent on downstream boundary conditions, which can be negligible in general.

The *Industrial Silencing Handbook* [12] defines dynamic insertion loss (DIL) as the numerical difference in dB between two sound pressure levels measured at the same location, under controlled flow conditions, before and after installation of a silencer. To measure the insertion loss of the discharge silencer would require that the compressor first be operated with a discharge piping system that did not include the silencer. Pulsation (noise) data would be measured in the discharge piping at a location which would be downstream of the silencer, if the silencer was installed. Next, the silencer would be installed in the discharge piping and a second set of pulsation data would be obtained at the test location downstream of the silencer with the compressor operating at the same conditions. The insertion loss would be the difference in sound pressure levels between the two sets of data. Since compressors are rarely operated without a discharge silencer, it is difficult to actually measure the insertion loss.

Silencer manufacturers' catalogs usually provide noise attenuation curves for various silencers. The attenuation values in dB are usually plotted versus the octave band center frequencies from 63 Hz to 8K Hz. The curves represent insertion loss of airborne noise for typical applications under average conditions. Since it is difficult to accurately predict the expected performance of a silencer over a wide range of applications and conditions, the curves should be used with discretion.

The insertion loss values in each octave band for the inlet and discharge silencers previously recommended in an earlier edition of API 619 are shown in the following table.

**Insertion Loss (dB) for
Octave Band Center Frequencies, Hz**

Freq, Hz	63	125	250	500	1K	2K	4K	8K
Inlet	20	20	20	20	20	20	20	20
Discharge	30	30	30	30	30	30	30	30

It is impractical to measure the insertion loss for silencers; however, the insertion loss can be estimated using computer simulations. The computer model should include the compressor discharge port, the discharge nozzle, the discharge silencer, and the discharge piping. The model is excited using unity pulsation amplitude at all frequencies from 1 Hz to 10,000 Hz. The pulsation levels can then be computed at several locations in the piping system. The insertion losses should be computed for the full range of operating conditions.

An example of computed insertion losses is shown in Figure 17 for natural gas, nitrogen, and process gas.

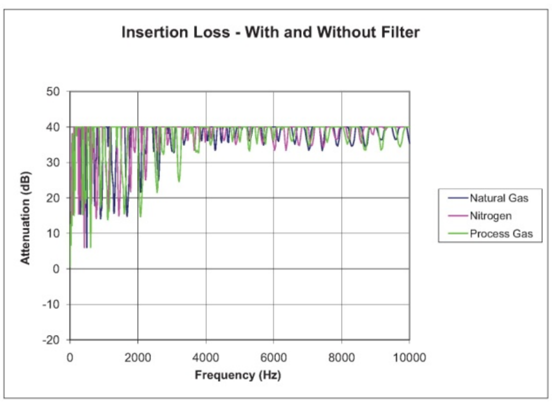


Figure 17: Insertion Loss - With and Without Filter

Although the program assumes that the pulsations are one-dimensional plane waves, the computed acoustical natural frequencies should be accurate for responses up to approximately 400 Hz (first crosswall mode) for the silencer used in this example. The minimum insertion losses occur at the valleys (dips) in the plots which correspond to the acoustical natural frequencies. As shown, many acoustical natural frequencies are excited. The reduced attenuation near 50 Hz is due to the amplification of the silencer Helmholtz frequency. In the higher frequency ranges between 200 Hz and 2000 Hz which would include the multiples of the PPF, the computed insertion losses are approximately 15 dB. These insertion losses for this example are well below the recommended minimum insertion loss of 30 dB.

Pulsation Reduction for Given Insertion Loss

For sound pressure, the relationship between differences in sound pressure (which is the insertion loss measurement) and the reduction in pulsation (P) is defined as follows:

$$dB_1 - dB_2 = 20 \log (P_1/P_2) \qquad (4)$$

where:

$dB_1 - dB_2$ = insertion loss (DIL) provided by the silencer

P_1 = inlet pulsation amplitude

P_2 = outlet pulsation amplitude

The above equation can be re-written to calculate the reduction in pulsation for a given insertion loss

$$(P_1/P_2) = \log^{-1} (DIL/20) \qquad (5)$$

The linear attenuation factor for various insertion losses in dB are shown in the following table.

DIL, dB	Linear Attenuation Factor
3	1.4
6	2.0
10	3.2
15	5.6
20	10.0
25	17.7
30	31.6
35	56.2
40	100

Pulsation levels at the compressor discharge flange often exceed 20 percent of the static pressure, especially when the compressor is operating in an over-compression or under-compression condition. In these cases, the required insertion loss could be greater than 30 dB, which illustrates why the insertion loss for the discharge silencer should be at least 30 dB or greater.

SHOP TESTS

Dry screw compressors are normally run in the manufacturer's shop to verify that the compressors are correctly assembled and that the rotors mesh correctly without rubbing throughout the operating speed and load range. Shaft vibrations are usually monitored during these tests. These tests are referred to as "Mechanical Running Tests" and are described in API Standard 619.

During these tests, the compressor is usually operated on air without the silencers. The discharge is vented to the atmosphere. In addition, the compressors are often operated at reduced speed due to the higher molecular weight of air. The pulsations and noise levels are usually not monitored during these tests because the silencers are not installed and the compressor may not be running near the design conditions.

For critical units, many companies require that the screw compressors be tested with the contract silencers and contract gas. If the unit cannot be run in the shop with the contract-gas, then closed loop tests are performed using a non-flammable test gas which has a molecular weight similar to the contract-gas. Helium is often used for low molecular weight contract-gases and nitrogen or various refrigerant gases are used for high molecular weight contract-gas [5]. The test gases are often blended in an attempt to obtain thermophysical properties (molecular weight and speed of sound values) similar to the contract-gas.

During these tests, the compressor is usually driven with a variable speed driver, such as an electric motor or a steam or gas turbine, which allows the compressor to be tested over a full range of operating conditions.

It is important that the actual contract suction and discharge silencers be installed during the test. Detailed vibration, pulsation, and noise data should be acquired during these tests. The tests are similar to field tests which are described in the following section.

FIELD TESTING

Field testing is generally required to diagnose pulsation, vibration, and noise problems associated with screw compressors. The test program philosophy is to try to identify energy generation mechanisms (such as pulsation at multiples of the PPF) and determine transmission paths and/or amplification mechanisms. The following is a summary of the typical instrumentation and test procedures. This information is discussed in more detail in a paper previously presented by the author [19].

Pulsation

High-frequency pulsation data are usually acquired with piezoelectric pressure transducers. The pressure transducers should be rated for the service with regard to the pressure, temperatures, and gas composition (H_2S). The transducers should be vibration compensated to avoid false signals due to the vibration of the transducer.

Pressure pulsation data should be acquired at several locations in the system, such as the discharge nozzle, silencer, and downstream piping. Due to the lack of pressure test locations, pressure transducers are often installed in the available vents and drains connections. These locations are often referred to as "stubs" because the acoustical ¼ wave stub resonances can be excited. In many cases, the pulsation data obtained at these locations will not be accurate due to the excitation of these stub resonances [19]. To avoid the problems with the stubs resonances, the pressure transducers can be installed using insertion probes that allow the transducer to be inserted through a valve and positioned inside the pipe wall. The transducers can also be mounted directly into the wall of the pipe; however, this will require depressurizing the system to remove the transducers.

Static Pressure

Static pressure transducers should be installed to measure the system operating pressures.

Shell Wall Vibration

The measurement of piping shell wall vibration requires low-mass, high-frequency accelerometers. The accelerometers should be screwed to pads that are either glued or welded to the structure. In some cases, acceleration levels in excess of 500 g's 0-peak have been measured; therefore, the accelerometer sensitivity should be low (10 mv/g) to prevent overloading the accelerometer. Such extreme vibration amplitudes can damage accelerometers, or cause the electrical connections and wires to fail.

The vibration data should be recorded in acceleration units (g's) or velocity units (in/sec). The allowable shell wall vibration to avoid fatigue failures is approximately 6 in/sec peak for carbon steel piping [9, 19]. These allowable levels were developed for evaluating high-frequency shell wall type vibration and should not be used for evaluating low-frequency lateral type piping vibration. These allowable shell wall vibration levels were computed using a stress concentration factor (SCF) of 5.

Strain Measurements

Strain gages can be used for evaluating the system with regard to possible fatigue failures. The strain gages are typically installed at high-strain locations on the discharge nozzle, expansion joints, silencer, downstream piping, and small-bore piping. The following criteria can be used to evaluate strain levels for carbon steel piping. These strain criteria are equivalent to an allowable stress of 3000 psi and conservative values for the SCF and fatigue strength of the material [19].

Dynamic Strain Criteria for Carbon Steel Piping

Strain Level, micro-strain peak-peak	Comment
Less than 100	Safe
100 – 200	Marginal
Greater than 200	Excessive

Sound Pressure Level Measurements

Sound level meters are generally used to measure the sound pressure ("noise") levels near the compressor and the piping. When evaluating the sound pressure levels with regard to human exposure, the levels are measured using the A weighting scale (dBA). However, when using sound pressure level measurements to determine the relationship between the sound, and the vibration and noise data, the C or flat weighting scale (dBC) should be used.

The sound pressure levels are normally measured approximately 3 ft. from the screw compressor/piping system. The sound pressure levels can also be used to estimate the piping shell wall dynamic strain levels by measuring 1 inch from the shell wall. Sound pressure levels of 136 dBC measured at approximately 1 inch from the pipe wall are equivalent to dynamic strain levels of approximately 200 micro-strain peak-peak which would be sufficient to expect fatigue failures in carbon steel piping.

Rotor Vibration

The compressor rotor vibrations should be measured using proximity probes as discussed in API Standard 619. [5]

Impact Tests

The shell wall mechanical natural frequencies and vibration mode shapes of the silencer and discharge piping can be measured using impact tests. These tests should be performed when the compressor is not running and the background vibration levels are low [19]. It should be determined if the measured shell wall natural frequencies could be excited by energy at multiples of the PPF.

Test Procedure

The compressor should be operated over the full range of operating conditions (speed, pressures, temperatures, and molecular weights of the gas). The test data (rotor vibration, shell wall vibration, structural vibration, pulsation, static pressures, strain, and sound pressure levels) should be continuously recorded during the tests. The unit operating conditions during the tests should also be obtained from the plant DCS system.

COMPUTER SIMULATIONS

Computer analyses are often required to evaluate the acoustical and mechanical characteristics of screw compressor/piping systems. These analyses can be performed in the design stage to help avoid potential problems, or after the system has been installed to determine the causes of the problems and to evaluate possible modifications to correct the problems.

Dry Screw Compressors

The analyses would include pulsation analyses of the compressor/discharge silencer/discharge piping system, and mechanical analyses of the discharge silencer.

Wet Screw Compressors

The analyses would include: pulsation analyses of the compressor/oil separator system, pulsation analyses of the oil injection system, and mechanical analyses of the oil separator.

Pulsation Analyses

Following the field tests, computer simulations can be made to compute the acoustical natural frequencies and pulsation levels throughout the system (compressor discharge nozzle, discharge silencer, and downstream piping). The computer model would be normalized to match the actual field data. After the computer model matches the field data, then possible modifications would be evaluated to determine the optimum modifications to reduce the pulsation levels.

It is difficult to compute the flow modulation (pulsation) generated by screw compressors. Algorithms for computing the flow modulation in reciprocating compressors have been available for many years; however, similar algorithms are not currently available for screw compressors. As previously discussed, there are many variables which can affect the pulsation levels generated by screw compressors, such as, the internal geometry of the compressor, internal clearances, over-compression, under-compression, etc.

Many screw manufacturers do not know the exact flow modulation generated by their compressors and often assume that the flow modulation is approximately 10 – 20 percent of the average flow. API 619 states that the allowable pulsation levels in the piping downstream of the silencers should be limited to approximately 2 percent peak-peak of the discharge pressure; however, it is often difficult to achieve these pulsation levels when the pulsation generated by the compressor are not accurately known.

Mechanical Analyses

Finite element models (FEM) are often made to compute the mechanical natural frequencies of the silencer.

DESIGN GUIDELINES

The following is a list of suggested guidelines regarding the selection and operation of screw compressors. These guidelines are generally applicable for both dry and wet screw compressors.

Compressor Internal Volume Ratio

The built-in or internal volume ratio (V_i) is a fundamental design characteristic of all screw compressors. The discharge pulsation levels are minimized when the internal pressure is approximately equal to the discharge line pressure. Therefore, the compressor internal volume ratio should match the compressor operating conditions. For installations with large variations in loading, it is often better to have several smaller compressors rather than one or two large compressors, since this arrangement allows the correct match between the operating conditions and the compressor design conditions to be more easily maintained.

Vessel and Pipe Wall Thickness

The shell wall vibration levels and the resulting noise levels can be significantly reduced by using thicker wall piping. The wall thickness should not be simply based on the static pressure or corrosion requirements which often indicate that thinner walls would be acceptable. In addition, as stated in API Standard 619, the wall thickness should not be reduced when using non-carbon steel piping. The author recommends that the wall thickness on discharge silencers and oil separators should be at least 1 inch thick.

Discharge Silencer Design

The silencers should be designed to attenuate high-frequency pulsation up to a frequency of approximately 10x PPF. This requirement necessitates that no acoustical natural frequencies should exist within the silencer itself below 10x PPF.

The discharge silencer insertion loss should be at least 30 dB for compressors that operate with minimum pulsation levels where the internal pressure matches the discharge pressure. The silencer insertion loss should be increased to approximately 40 dB to attenuate the higher pulsation levels when the compressor is operating in under-compression and over-compression conditions.

Pulsation analyses should be performed to compute the acoustical natural frequencies and pulsation levels for the compressor/discharge silencer/discharge piping system. The analyses should consider both the 1-dimensional and 3-dimensional acoustical natural frequencies.

The pressure drop values specified in API Standard 619 5^{th} Edition [5] should be used as a guideline. In some cases with high pulsation levels, the allowable pressure drop may need to be increased to attenuate the pulsation to acceptable levels.

In critical applications, finite-element-analyses (FEA) should be performed to compute the mechanical natural frequencies of the silencer and the downstream piping.

Orifices

As discussed, the orifice thickness on screw compressors should be a minimum of approximately 0.5 – 1.0 inches to avoid possible fatigue failures. It is often difficult to install the thick orifices in the piping system after the system is constructed. Therefore, it is recommended that the piping system be designed with thick spacers (approximately 1 inch thick) at the compressor discharge flange and at the discharge silencer outlet flange in the event that orifice plates have to be added at a later date.

Discharge Piping Pulsation

For typical systems found in industrial applications, experience has shown that pulsation levels of approximately 2 percent peak-peak of the discharge pressure generally results in acceptable piping vibration levels. This criterion also agrees with the allowable levels specified in the API Standard 619 current Edition.

Noise Treatment

High-frequency sound insulation should be installed on the discharge silencers and the downstream discharge piping. Similar noise treatments should also be installed on oil separators.

Screw compressors also radiate high-frequency noise due to the structural vibration of the compressor body and the gear housing on dry screw compressors. It would be difficult to attenuate the noise generated by the compressor body; however, in some cases it may be possible to install noise treatments on the gear housing. The compressors are often installed inside noise enclosures in an effort to reduce the noise levels but the enclosures also create other problems, such as overheating and maintenance problems due to reduced accessibility to the compressor.

Noise Levels

It is desirable that the sound pressure levels near screw compressors be less than 85 dBA, which is the noise level often quoted in users' purchase specifications. However, in many instances, the background levels exceed 85 dBA without the compressors running. The author's experience indicates that it is very difficult, if not impossible, to attain these low sound pressure levels. Therefore, it is suggested that the allowable sound pressure levels near screw compressors be increased to levels there are more attainable, such as 90 – 95 dBA.

CASE HISTORY NO. 1

Dry Screw Compressor
Damage to Rotors Due to Orifice Failure

A dry screw compressor was installed in a vapor recovery unit and operated at a constant speed of 4060 RPM (1x PPF = 271 Hz). Pulsation data obtained during the shop tests had shown high pulsation levels at multiples of the compressor pocket passing frequency of 271 Hz. The maximum amplitude was approximately 104 psi peak-peak at 1x PPF with significant pulsation at 6x PPF.

A pulsation analysis indicated that pulsation at the PPF was amplified by an acoustical natural frequency of the compressor discharge nozzle and the silencer inlet nozzle. The analysis also indicated that the pulsation at this acoustical natural frequency could be reduced by a factor of 5 or more by installing an orifice plate with a pressure drop of 1 percent at the compressor discharge flange.

The orifice plate was designed and fabricated using the user company's recommended design procedures. Although no vibration, pulsation, or noise data were obtained after the orifice was installed, personnel at the site reported that the vibration of the silencer and the discharge piping were reduced, and the noise levels near the discharge silencer were also reduced.

After operating for approximately 2 months, the orifice plate experienced a fatigue failure and a broken part from the orifice entered the compressor causing extensive damage to the compressor rotors (Figure 18). The discharge silencers were an "up-connected" configuration. It was theorized that the broken orifice parts remained suspended in the discharge silencer when the compressor was in operation and then fell downward into the compressor rotors when the unit was stopped. The damage occurred when the compressor was restarted.

Figure 18: Orifice Fatigue Failure

Analyses were performed in an effort to determine the possible causes for the orifice failure. A metallurgical analysis verified that the failure was a fatigue failure. A finite-element analysis (FEA) of the installed orifice plate indicated that the orifice fatigue failure could have been caused by excitation of a mechanical natural frequency of the orifice plate at approximately 1636 Hz. Possible excitation sources were pulsation at 6x PPF (1626 Hz) or vortices generated by flow across the orifice plate. The pulsation at 6x PPF was considered to be the most likely cause since significant pulsation at 6x PPF was measured during the shop tests.

The finite element analysis showed that the mechanical natural frequencies of the orifice plate could be raised by increasing the thickness of the orifice plate. The strength of the orifice plate could also be increased by using a flat plate without the tapered or beveled section.

In addition to raising the natural frequencies, increasing the orifice plate thickness also decreases the stresses for a given level of vibration. For example, increasing the plate thickness from 1/4 inch to 7/16 inch causes the bending stress to be reduced by a factor of three. However, there is a practical limit to the thickness of the orifice plate due to alignment problems with the piping.

Therefore, it was recommended that the orifice plate thickness be increased to 7/16 (0.4375) inches. Also, the plate should not be beveled or tapered, as is often done with orifice plates in flow meters. Although the analysis indicated that increasing the orifice thickness would prevent future fatigue failures, the operating company decided not to replace the orifice plate but to replace the original discharge silencer with a new silencer that was designed to attenuate the pulsations to meet the API 619 requirements.

CASE HISTORY NO. 2

Dry Screw Compressors
Fatigue Failures of Discharge Silencers

There were three dry screw compressors operating at this plant.

Service	HP	Operating Speed	Pressure Ratio
Reducing Gas	5476	3780 RPM	1.8
Cooling Gas	1261	3100 RPM	1.25
Transport Gas	2013	5750 RPM	4.25

The three compressors had inlet and discharge silencers which were mounted vertically on the top of the compressors. The silencers were reactive type with two chambers connected with choke tubes and were similar to the designs shown in Figure 12. The silencers did not have any absorption material.

All of the discharge silencers experienced excessive pulsation, vibration, and noise levels when the compressors were initially started on nitrogen. The excessive pulsation and vibration levels resulted in extensive fatigue failures on all the discharge silencers – the Reducing Gas Compressor failed in less than four hours, the Cooling Gas Compressor failed in less than six hours, and the Transport Gas Compressor failed in approximately 30 hours. Due to the fatigue failures, all three discharge silencers had to be taken out of service. One of the silencers had a fatigue crack in the shell that propagated almost the entire length of the silencer (Figure 19).

Figure 19: Discharge Silencer Fatigue Failure

In an effort to maintain production, the failed silencers were removed and replaced with straight sections of pipe with no internal baffles or choke tubes (Figure 20). The pipes were the same diameter as the compressor discharge nozzles.

Figure 20: Discharge Silencer Removed and Replaced with Pipe

The silencer manufacturer designed a resonator (smaller diameter stub) which was welded to the side of the 20-inch pipe on the Reducing Gas Compressor. The short stub is referred to as a side-branch resonator and was tuned for a frequency which matched 2x PPF. This resonator was designed to attenuate the pulsation at 2x PPF in the 20-inch pipe.

Field tests were performed to obtain pulsation, vibration, strain, and noise data on the compressors during operation with the straight pipes. The following is a list of the major conclusions.

1. The measured pulsation, vibration, and noise occurred at multiples of the PPF. The measured vibration and noise levels were considered to be excessive. Piping shell wall vibration levels up to 300 g's 0-peak were measured. (Due to the high vibration, special metal mounting pads were welded to the piping and the accelerometers were then screwed to the pads.) The maximum sound pressure levels measured near the discharge piping were approximately 125 dB and could be heard outside the plant.
2. During the tests of the Reducing Gas Compressor, several fatigue failures occurred on the pressure taps and tubing. The resonator installed to attenuate the pulsation at 2x PPF also failed and actually fell off the piping.
3. Pulsations generated by the compressors were amplified by acoustic cross-wall natural frequencies of the piping.
4. Piping vibration levels were amplified when the pulsation frequencies were coincident with the piping shell-wall mechanical natural frequencies.
5. Pulsation occurred at multiples of the PPF. The predominant amplitudes shifted between the various multiples as the process varied. The variation in the operating conditions caused the speed of sound of the gas to change which shifted the acoustic cross-wall natural frequencies.
6. Although no data were acquired on the original silencers, calculations indicated that the silencers had cross-wall natural frequencies which could amplify the pulsation generated by the compressors.

Since replacement silencers would not be available for several months, the plant decided to install multi-hole orifice plates in an attempt to attenuate the excessive pulsation levels. The tests indicated that the orifice plates attenuated the pulsation levels, which in turn reduced the vibration and noise to acceptable levels.

Although the orifice plates were effective in attenuating the pulsation levels, the pressure drop across the plates was excessive. The pressure drop across one of the plates was so high that the motor tripped on excessive current during startup. The orifice plate had to be redesigned to reduce the pressure drop to allow the compressor to operate. Although the pressure drop was reduced, the pressure drops across the orifice plates on all three compressors were well above the API allowable level. The excessive pressure drops resulted in increased horsepower requirements which in turn increased electrical costs.

The plant considered the orifice plates to be a temporary solution. The final solution was to design silencers which would operate satisfactorily without experiencing excessive pulsation, vibration, and noise problems.

CASE HISTORY NO. 3

Dry Screw Compressors
Excessive Noise after Replacing Silencers & Piping

Five dry screw compressors were installed at this installation to collect vapors from oil storage tanks and flue gases from the boilers. The molecular weight of the gas varied from approximately 28 to 45 depending upon the mixture of stock tank vapors and flue gas. The compressor inlet operated at a vacuum with a design pressure of 12.5 psia. The discharge pressures varied from 28 to 40 psia.

After several years in service, the gases caused extensive corrosion damage to the suction and discharge silencers, and the suction and discharge piping. Consequently, the silencers and the piping were replaced. As described below, the replacement silencers and piping were significantly different than the original system, and the carbon-steel piping was replaced with stainless steel piping with thinner walls and no insulation.

Comparison of Original and Modified Piping Systems

Item	Original System	Modified System
Suction Piping	24-inch carbon steel (0.5 inch wall) Insulated – rock wool, metal lagging	24-inch stainless (0.195 inch wall) No insulation
Suction Silencer	60-inch dia (0.625 inch wall) Insulated – 2 inches of fiber, metal lagging	30-inch stainless vertical pipe No insulation
Discharge Silencer	36-inch dia (0.75 inch wall) Insulated – 3 inches of fiber, metal lagging	36-inch dia (0.75 inch wall) No insulation
Discharge Piping	18-inch carbon steel (0.5 inch wall) Insulated – rock wool, metal lagging	18-inch stainless (0.19 inch wall) No insulation

Each compressor was installed in a separate sound enclosure. All five compressors were installed inside a large building. In the original design, each compressor was equipped with reactive suction and discharge silencers similar to those shown in Figure 12.

The compressors had operated with acceptable noise levels for several years with the original piping system. The sound pressure levels inside the compressor building were less than 90 dB with several compressors in service. However, the sound pressure levels in the building significantly increased with the modified piping system. Typical sound pressure levels with only one compressor in service were approximately 105-109 dBA inside the compressor building and 98-102 dBA outside the building.

The shell wall vibration of the discharge piping caused the noise levels to be increased outside the compressor building. The high noise levels outside the compressor building were particularly objectionable because the discharge piping was routed near an office building which caused the noise levels inside the office building to also be increased.

The noise levels of one of the compressors were significantly louder than the other units and the noise could literally be heard several miles away. This compressor had been recently overhauled and all of the internal clearances were returned to the original clearances which improved the compressor performance; however, the discharge pulsation levels were significantly increased. Due to the high noise levels, the user company preferred not to run this unit.

Most of the changes (removing the suction silencer, changing the design of the discharge silencer, replacing the piping with thinner wall piping, and removing the insulation) contributed to increasing the noise levels. The user company felt that the noise levels could probably be attenuated by returning the suction and discharge piping to the original configuration; however, these modifications were considered to be too costly.

The maximum noise occurred in a frequency range of 680 – 1360 Hz at 4x, 5x, 6x, 7x, and 8x PPF. The noise level at the fundamental PPF of 168 Hz was considered to be low.

Pulsations generated by the compressor at multiples of the PPF were amplified by acoustic natural frequencies of the compressor/silencer/piping system. Additionally, pulsation amplitudes and noise at the higher multiples of the PPF varied considerably every few seconds. The variation in pulsation levels appeared to be due to the slight changes in the molecular weight of the gas which shifted the speed of sound of the gas mixture. The pulsation amplitudes were generally lower when the compressors were operating with higher levels of flue gas.

In the discharge silencer, the maximum pulsation levels were approximately 25 psi peak-peak at 5x PPF (840 Hz) and 15 psi peak-peak at 6x PPF. Downstream of the discharge silencer, pulsation amplitudes were approximately 5-7 psi peak-peak at several multiples of the PPF (5x, 6x, 7x, and 8x), which were significantly higher than the API allowable levels.

Piping vibration levels were amplified by excitation of the shell wall natural frequencies resulting in excessive noise levels. The piping shell wall vibration levels were not considered to be excessive with regard to fatigue stress levels. There were no reports of structural cracks, or failures of attached fittings or instrumentation. Therefore, the primary problem was environmental noise which was considered to be excessive to the personnel at the facility.

Since the vibration levels were not considered to be excessive, the most economical modification to reduce the excessive noise levels was to install sound barrier jacketing in combination with several inches of insulation on all of the suction and discharge piping. These modifications were similar to those discussed in the paper previously presented by the author [19]. It was reported that these modifications attenuated the noise levels to acceptable levels.

CASE HISTORY NO. 4

Dry Screw Compressor
New Silencer Design

This dry screw compressor was installed in a "tail gas" service in a refinery and operated at 7883 RPM (1x PPF = 525 Hz). The system had experienced excessive high-frequency vibration of the discharge silencer and the downstream piping which resulted in fatigue failures of the attached small bore piping instrument lines. The tail gas was primarily a mixture of hydrogen sulfide, hydrogen and methane. Downtime associated with failures was extremely costly.

Based upon data acquired over a four-week period, it was determined that traditional silencer designs would not be effective at reducing the vibration. Therefore, a novel silencer design was suggested.

The silencer was designed to limit critical dimensions but achieve the necessary acoustical behaviors to attenuate the pulsations and the vibrations of the silencer and the attached piping. Internal absorptive material was not employed. A cutaway sketch of the new silencer is shown in Figure 21.

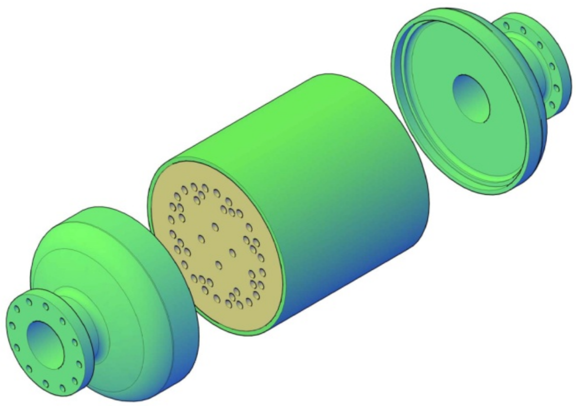


Figure 21: Multi-Pass Flow Silencer

The new design provided excellent attenuation of not only the lower PPF's, but also of the especially harmful higher frequencies (up to 10x PPF). Vibration data were obtained before and after the installation of the new silencer.

Data were obtained at several locations on the silencer and piping. For ease of comparison, frequency spectra from all of the locations are overlaid on one plot to create an overall composite of all of the data. The data with the original silencer are plotted in Figure 22 and the data with the new multi-path silencer are plotted in Figure 23. As shown, the vibration levels were significantly reduced with the new silencer. The maximum vibration levels at 2625 Hz (5x PPF) were reduced from approximately 220 g's peak-peak to 20 g's peak-peak. Vibration throughout the compressor system piping, adjacent compressor, and associated structures was also significantly reduced.

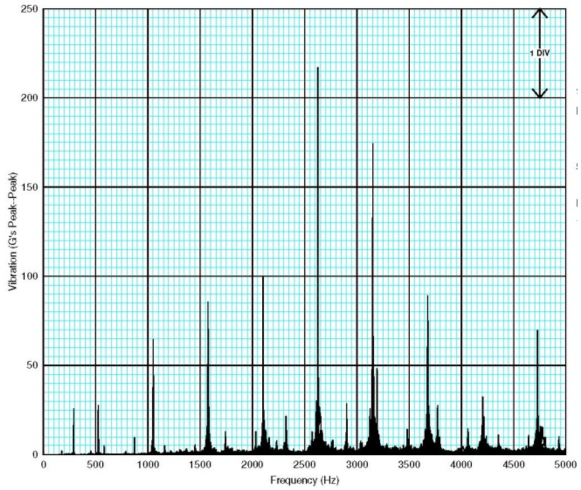


Figure 22: Vibration Summary with Original Silencer

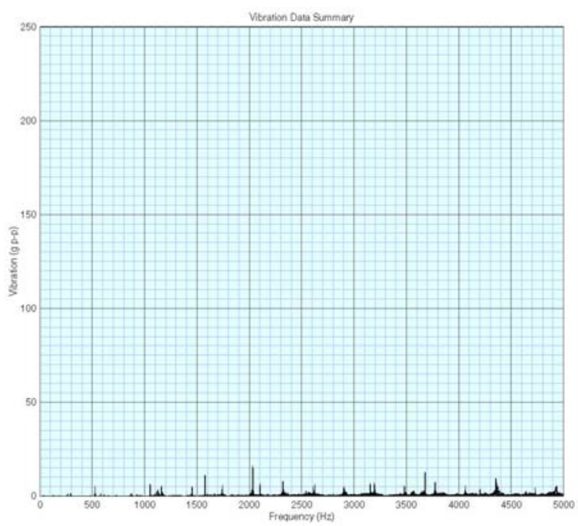


Figure 23: Vibration Summary with New Silencer

CASE HISTORY NO. 5

Wet Screw Compressor Fatigue Failures of Lube Oil Piping

An ammonia (NH_3) refrigeration compressor experienced fatigue failures of the compressor lube oil piping. In addition, the noise levels near the vertical oil separator were approximately 115 – 120 dB which were excessive for the operating personnel and double-hearing protection was required.

The compressor was driven by a 3500 HP electric motor at 3600 RPM. The screw compressor had 5 lobes on the male rotor which resulted in a PPF of 300 Hz.

Field tests were performed to determine the causes for the excessive noise and vibration levels. As shown in Figure 24, strain gages were installed at several locations where vibration induced failures were possible. **Note the paper should be "zoomed" to locate the test points.*

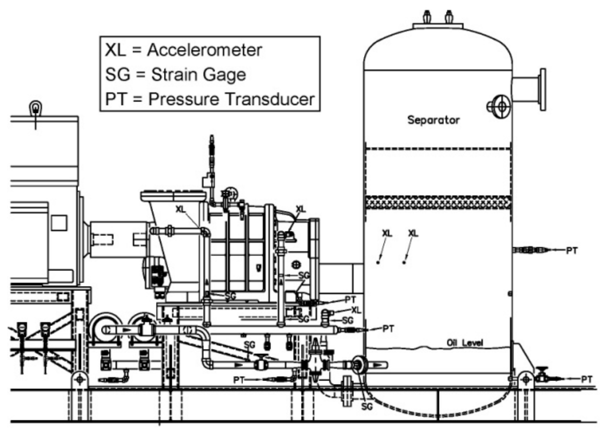


Figure 24: Test Locations

Similarly, accelerometers were attached to the lube-oil piping and the separator. Pulsation data were also acquired in the compressor discharge piping, oil separator, and the compressor lube-oil piping. Data were obtained as the slide valve position was changed from 0-100 percent.

Lube Oil Piping Failures

The data indicated that the fatigue failures of the compressor oil injection lines were due to excessive vibration caused by high level pulsation at the PPF. It was theorized that the source of the pulsation in the lube oil lines was a "chopping" of the oil flow into the oil gallery by the rotation of the compressor rotors.

A pulsation analysis was performed to evaluate possible modifications to attenuate the pulsation levels in the lube oil lines. The analysis indicated that the pulsations were amplified by an acoustical natural frequency of the lube oil piping system near the PPF. The analysis also showed that the pulsations could be significantly reduced by shifting the acoustical natural frequency away from the PPF.

Based on the analysis, a volume bottle was installed in the lube oil line near the compressor oil port (Figure 25).

Figure 25: Bottle Added to Reduce Pulsation in the Lube Oil Piping

Data obtained after the bottle was installed indicated that the pulsation, vibration, and strain levels were significantly reduced to acceptable levels. Frequency spectra of the oil pulsation in the original and modified condition are shown in Figure 26.

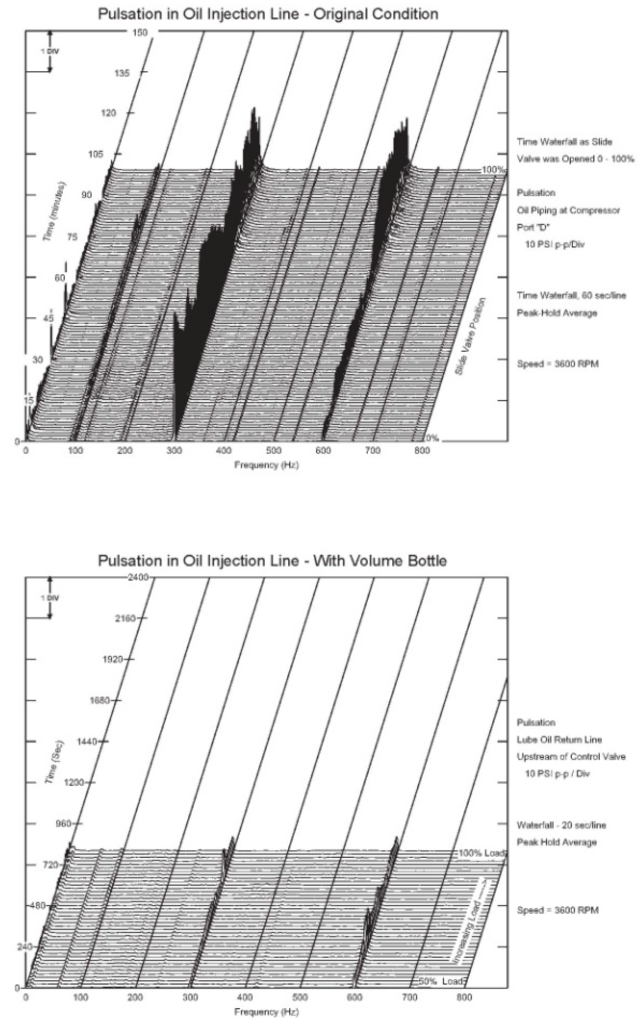


Figure 26: Frequency Spectra in Oil Injection Line

Separator Vibration and Noise

The field measurements indicated that the excessive noise levels (sound pressure levels) were due to the separator shell wall vibration at the PPF.

PPF pulsation levels inside the separator were higher than the levels in the compressor discharge piping, which indicated that the pulsation levels were being amplified inside the separator. Calculations verified that PPF pulsations were amplified by an acoustical cross-wall natural frequency (2-diameter mode) at 300 Hz.

The best method to eliminate the cross-wall natural frequencies in the separator would have been to install vertical baffles in the separator. However, this modification would have required disassembly of the separator which would have required a long time to implement and would have been too disruptive to the plant operation.

Pulsation analyses were performed to investigate other methods to reduce the pulsation levels inside the separator. One method would be to install an orifice at the inlet to the separator. However, there was concern that the orifice plate could experience erosion damage due to the entrained oil. Therefore, it was decided not to install an orifice plate.

One member of the team suggested adding resonators to the compressor discharge piping upstream of the separator. Such solutions are not usually favored because of structural vibration issues that can occur on the resonators. (Recall that a similar resonator in Case History 2 failed due to excessive vibration.) However, the computer analyses indicated that resonators would reduce PPF pulsation levels in the separator.

The resonators consisted of three short sections of piping (stubs) which were designed such that the ¼ wave acoustical natural frequency of the stub was equal to the PPF. Three holes were cut into the discharge pipe and the resonators were welded to the pipe. The resonators were oriented such that the oil would drain back into the discharge pipe (Figure 27).

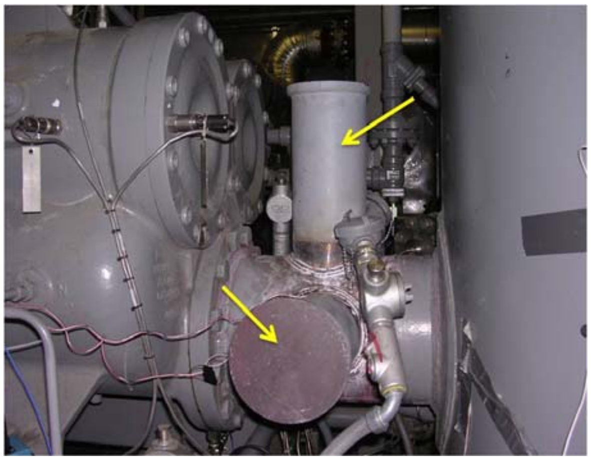


Figure 27: Resonators Added to Reduce Pulsation in the Separator

As a precaution, prior to starting the compressor, impact testing of the resonators was done to determine the structural natural frequencies of the installed resonators. The data showed that the resonators had a structural natural frequency at the PPF (300 Hz) which was coincident with the PPF, Figure 28.

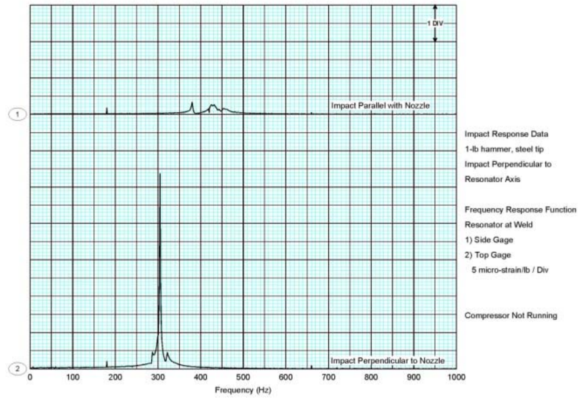


Figure 28: Impact Test on Resonators Without Gussets

The structural mode shape of the resonators indicated that gusset plates installed between the resonators would detune the structural response.

With the gussets installed (Figure 29), the natural frequency shifted away from the PPF to 420 Hz, which was an adequate separation margin (Figure 30).

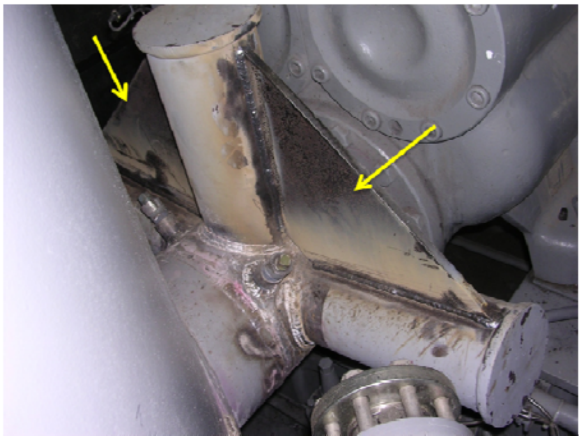


Figure 29: Gussets Installed

Data obtained with the resonators installed showed that the pulsations in the discharge piping and inside the separator were significantly reduced. Pulsation data measured before and after the installation of the resonators are shown in Figure 31.

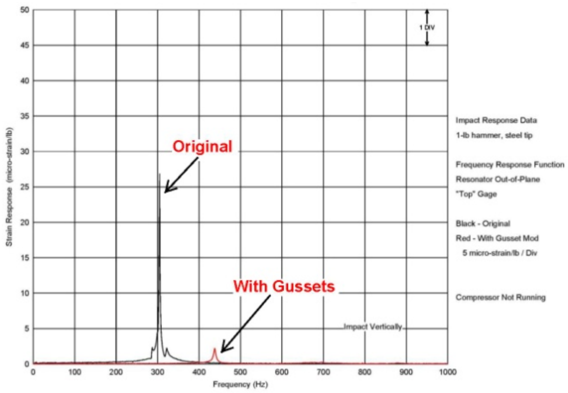


Figure 30: Impact Test on Resonators With Gussets

The vibration levels on the separator were also reduced. The noise levels were reduced from approximately 120 dB to between 100 and 108 dB which is approximately a factor of 4.

The resonators were considered to be a permanent modification and no further modifications were installed. Further noise reduction would require noise insulation on the separator.

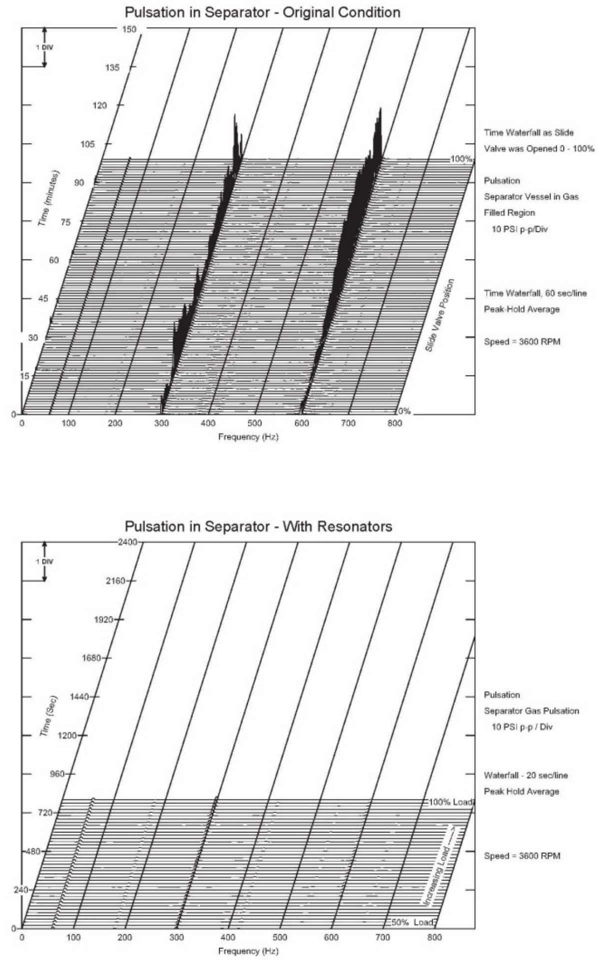


Figure 31: Pulsation in Separator

CASE HISTORY NO. 6

Wet Screw Compressor
Incorrect Internal Volume Ratios

Four CO_2 refrigeration compressors experienced high-level vibration on the oil separators which resulted in fatigue failures and excessive noise levels. The compressors were driven by electric motors at a constant speed near 3600 RPM. The compressor male rotor had 4 lobes (PPF = 240 Hz).

Compressors C-1, C-2, and C-3 were directly mounted on top of the horizontal oil separators, such as shown in Figure 32.

Figure 32: Compressor Mounted on Top of Oil Separator

The C-4 compressor was mounted on the ground adjacent to the oil separator. Test data indicated that shell wall vibration and noise levels near the oil separator were primarily at 240 Hz (1x PPF). In an effort to determine the cause(s) for the excessive vibration and noise levels, the shell wall vibration, noise (sound pressure level – SPL), pulsation in the 1st chamber, and pulsation in the oil pump discharge at 240 Hz for each compressor are compared in the following table.

Vibration, Noise, and Pulsation at 240 Hz for the Four Compressors

Unit	Shell Vibration	Noise SPL	Pressure Pulsation	
			1st Chamber	Pump Discharge
C-1	0.7 ips 0-p	110 dBC	0.8 psi p-p	7 psi p-p
C-2	0.1 ips 0-p	107 dBC	9 psi p-p	17 psi p-p
C-3	3.7 ips 0-p	117 dBC	1.1 psi p-p	12 psi p-p
C-4	3.7 ips 0-p	114 dBC	1.4 psi p-p	47 psi p-p

As shown, the shell wall vibrations, noise levels, and pulsation levels were significantly different on each unit. The vibration sensitivities due to the internal pulsations (vibration / pulsation) were also different on each unit. For example, on the C-1 unit, pulsation levels of 0.8 psi p-p resulted in vibration levels of 0.7 ips, while similar pulsation of 1.1 psi p-p on the C-3 unit resulted in much higher shell wall vibration levels of 3.7 ips. Conversely, pulsation levels on the C-2 were 9 psi p-p, but shell wall vibration levels were only 0.1 ips at 240 Hz.

The data indicated that the separator shell wall vibration levels were being affected by pulsation inside the oil separator; however, there were other factors that also affected the vibration levels. It was thought that the shell wall vibration levels were being increased due to forces from the compressors which were being mechanically transmitted to the shell wall.

The field tests also showed that the pulsation levels were very sensitive to the system discharge pressure and that changes of only a few psi significantly affected the pulsation and noise levels. The compressors were also sensitive to the slide valve position. Therefore, it was suspected that the increased pulsation levels could be due to over-compression or under-compression conditions which occur when the compressor internal pressure does not match the system discharge pressure. The following table is a summary of the calculated and measured pressure ratios.

Comparison of Calculated and Measured Pressure Ratios

Unit	Pressure Ratio		Percent Difference	Comment
	Internal	Measured		
C-1	3.8	3.9	-3%	Near Design
C-2	3.5	5.2	-49%	Under-compression
C-3	4.5	4.0	11%	Over-Compression
C-4	2.6	2.2	15%	Did not operate at full load

As shown, C-1 was the only compressor that was operating near the design pressure ratio. In addition, C-1 experienced the fewest problems. C-2 was operating in an under-compression condition. C-3 and C-4 were operating in over-compression conditions.

These under and over-compression conditions were believed to be the primary causes for the increased vibration, noise, and pulsation problems. In addition to increasing the pulsation levels, the compressor shaft and compressor case vibrations were also increased.

Structure-borne vibration was transmitted to the separators, especially on the units where the compressor was mounted on top of the separator.

The vibration levels on the separators were further increased because several of the shell wall natural frequencies were near multiples of the PPF.

The immediate recommendations were to change the internal pressure ratios (V_i) of the compressors to match the system operating conditions and to install acoustic lagging on the separators. The long-term solution was to replace the separators with new separators with thicker walls which would reduce the shell wall vibration and the resulting noise.

CONCLUSIONS

1. Algorithms need to be developed for accurately computing the flow modulation (pulsation) generated by screw compressors at different operating conditions. Algorithms for computing the flow modulation in reciprocating compressors have been available for many years; however, similar algorithms are not available for screw compressors. It is difficult to accurately predict the pulsation levels, especially at the higher multiples of the PPF, in the silencer and downstream piping when the flow modulation produced by the compressor is not accurately known. Many compressor manufacturers do not know the exact flow modulation (pulsation) generated by the compressor and often use an approximate value, such as 10 – 20 percent of the discharge pressure, when evaluating silencer designs.

2. "Silencers" are installed to reduce piping pulsation and consequently structure-borne high-frequency vibration and resulting failures. Noise radiation from the compressor itself is often an equal contributor to other sources of sound radiation. Therefore, even well-designed silencers may not significantly reduce the ambient noise levels. Sound enclosures may be required to achieve acceptable environmental noise levels.

3. Silencers should be designed so that acoustical natural frequencies are not coincident with PPF excitation up to at least the 10^{th} order, and possibly higher on some units. The full range of possible mole weights (speed of sound) should be considered.

4. Structural natural frequencies of the silencers and oil separators should be well away from PPF excitation. Thicker shell-wall, internal reinforcing rings, and other such devices may be necessary. Damping material (constrained layer damping) may be necessary in situations where this design is not possible.

5. The screw compressors designs should comply with the specifications in API Standard 619 5^{th} Edition.

6. Full string tests should be performed for screw compressors installed in critical service.

NOMENCLATURE

c_p = specific heat at constant pressure

c_v = specific heat at constant volume

k = specific heat ratio

P_1 = inlet pulsation amplitude

P_2 = outlet pulsation amplitude

P_d = internal discharge pressure, psia

P_{dl} = discharge line pressure, psia

P_i = internal pressure ratio

P_s = internal suction pressure, psia

V_i = internal volume ratio

V_s = internal suction volume, acf

V_d = internal discharge volume, acf

REFERENCES

1. API Standard 619, 1st Edition, *Rotary-Type Positive Displacement Compressors for General Refinery Services*, 1975.

2. API Standard 619, 2nd Edition, *Rotary-Type Positive Displacement Compressors for General Refinery Services*, 1985.

3. API Standard 619, 3rd Edition, *Rotary-Type Positive Displacement Compressors for General Refinery Services*, 1997.

4. API Standard 619, 4th Edition, *Rotary-Type Positive Displacement Compressors for General Refinery Services*, 2004.

5. API Standard 619, 5th Edition, *Rotary-Type Positive Displacement Compressors for General Refinery Services*, 2010.

6. Briley, George, *Twin Screw Technology*, Technicold Services, Inc., www.gcbtechnicold.biz

7. Bruce, J. Trent, *Screw Compressors: A Comparison of Applications and Features to Conventional Types of Machines*, Toromont Process Systems, Calgary, Alberta, Canada.

8. Cashflo Limited, *The History of Screw Compressors*, www.cashflo.co.uk

9. EDI Staff, *Vibrations in Reciprocating Machinery and Piping Systems*, Engineering Dynamics Incorporated, Technical Report 41450-1, January, 2002.

10. Embleton, T. F., *Mufflers* in Leo Beranek, *Noise and Vibration Control*, Chapter 12, Institute of Noise Control Engineering, Cambridge, MA, 1988.

11. GE Power Systems – Oil & Gas, *Positive Displacement Compressors – Oil Free Rotary Screw*, A-C Compressor, Rev. 12/2001.

12. Golden, B. G. / Burgess-Manning, *Industrial Silencing Handbook*, Burgess-Manning, Inc., 1982.

13. Lovelady, Jeff and Bielskus, Peter, *High Frequency Fatigue Failures in Silencer/Pulsation Dampers for Oil-Free Screw Compressors*, Proceedings of the Twenty-Eighth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 141-145, 1999.

14. Mujic, Elvedin, et al, *Reduction of Noise in Screw Compressors*, 12th International Research/Expert Conference, *Trends in the Development of Machinery and Associated Technology*, TMT 2008, Istanbul, Turkey, August 26-30, 2008.

15. Murray, Kenneth, *What IS Noise?*, Facility Safety Management, http://www.fsmmag.com/Articles/2005/03/What_IS_Noise.htm

16. Nordquist, Glen and Blieskus, Peter, *Dry Screw Compressors in Process Gas Applications Including Maintenance Considerations*, Proceedings of the Twenty-First Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 3-19, 1992.

17. Ohama, Takao, et al, *Process Gas Applications Where API 619 Screw Compressors Replaced Reciprocating and Centrifugal Compressors*, Proceedings of the Thirty-Fifth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 89-96, 2006.

18. Pillis, J. W., *Basics of Operation, Application & Troubleshooting of Screw Compressors*, Frick Compressors, pp. 1-27, 1998.

19. Price, Stephen and Smith, Donald, *Sources and Remedies of High-Frequency Piping Vibration and Noise*, Proceedings of the Twenty-Eighth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 189-212, 1999.

20. *Rotary Movie – RG*, Ariel Corporation

21. Stosic, N., and Smith, I., A. Kovacevic, *Screw Compressors – Mathematical Modelling and Performance Calculation*, City University, School of Engineering and Mathematical Sciences, London, Springer-Verlag Berlin Heidelberg, 2005.

22. Wennemar, Jurgen, *Dry Screw Compressor Performance and Application Range*, Proceedings of the Thirty-Eighth Turbomachinery Symposium, Turbomachinery Laboratory, Texas A&M University, College Station, Texas, pp. 149-156, 2009.

BIBLIOGRAPHY

1. A-C Compressor, *Positive Displacement Compressors – Oil Free Rotary Screw*, 2001, GE Power Systems Oil & Gas, Appleton, WI.

2. A-C Compressor Corporation, *Preliminary Instruction Manual*, *Transmittal Information*, Oil Free Screw Compressors, 1997.

3. Burgess-Manning, *Brochure: Leaders in Sound Advice, Innovative Solutions For Over Sixty Years, Industrial Silencing, Gas/Liquid/Solid Separation, Pulsation Control*, Nitram Energy, Inc.

4. Diehl, George, *Machinery Acoustics*, John Wiley & Sons, New York, New York

5. Dowling, A. P., *Sound and Sources of Sound*, John Wiley & Sons, New York.

6. Eriksson, L. J., SAE Technical Paper Series, *Theory and Practice in Exhaust System Design*, Surface Vehicle Noise and Vibration Conference, 1985.

7. Eriksson, L. J., SAE Technical Paper Series, *A Review of Recent Progress in Exhaust System Design*, Earthmoving Industry Conference, 1982.

8. Evans, Jack, *Quiet that Compressor*, 2004, Plant Services.

9. Gardner-Denver, *CycloBlower – 9CDL Series Blowers, Parts List & Service Manual*, Cooper Industries.

10. Harris, Cyril, *Handbook of Acoustical Measurements and Noise Control*, McGraw Hill Publishing Company, New York, New York, 1991.

11. IRC TechNote, *Selection of Screw Compressors for Energy Efficient Operation*, 2002, Industrial Refrigeration Consortium, College of Engineering, University of Wisconsin-Madison, 2002.

12. MAN Turbo, *Process-Gas Screw Compressors (dry type) Compared with Reciprocating, Oil Flooded Screw, and Centrifugal Compressors*, MAN Turbomaschinen AG, 2003.

13. MAN Turbo, *Process-Gas Screw Compressors*, MAN Turbomaschinen AG, 2002.

14. MAN Turbo, *The Design, Selection and Application of Oil-Free Screw Compressors for Fuel Gas Service,* MAN Turbomaschinen AG, 2002.

15. Murray, Kenneth, *What is Noise? – Simple Attenuation with Silencing Equipment*, Facility Safety Management, www.fsmmag.com., 2005.

16. NORSOK (Norwegian Oil Industry Association) Standard R-100, *Mechanical Equipment Selection – Screw Compressors*, p. 9, 1997.

17. Reynolds, Douglas D., *Engineering Principles of Acoustics*, 1981, Allyn and Bacon, Inc., Boston.

18. Universal Silencer, *Rotary Positive Blower Silencers – Product Catalog No. 244-C*, Nelson Industries, Inc.

19. Universal Silencer, *Welcome to the World of Universal Solutions – 2005 Full-Line Catalog*

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